NASA TECHNICAL MEMORANDUM

NASA TM X-53707

February 16, 1968

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DRY FILM LUBRICATED BALL BEARINGS FOR GIMBALS OSCILLATING AT SMALL ANGLES IN VACUUM

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ABSTRACT

Present plans for the pitch and yaw gimbal system on the Apollo Telescope Mount (ATM) involve the use of flexible pivots. Because the development problems associated with a flexible pivot system operating at zero g. are unknown, it was decided to test dry lubricated ball bearings under simulated ATM conditions as a back-up to the flexible pivot system. Results of these tests indicate that dry lubricated ball bearings will operate satisfactorily for at least 240,000 cycles at a ±5° oscillation and at 120,000 psi Hertzian contact stress. The best lubricating system provides a maximum effective coefficient of friction per bearing of .0055 across 10° of oscillation.

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PROPULSION AND VEHICLE ENGINEERING LABORATORY
RESEARCH AND DEVELOPMENT OPERATIONS

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SUMMARY

Although the Apollo Telescope Mount pitch and yaw gimbal system primary design was based on flexible pivots, gimbal bearings were considered for the back-up design. The purpose of this investigation was to determine whether dry film lubricated ball bearings would have reasonable life when operating over small angles, only. Typical Teflon and glass sacrificial retainer ball bearings were evaluated with several different dry film lubricants on the bearing races. At operating conditions selected to simulate the subject application the bearings provided many times more than the required lifetime.

INTRODUCTION

Present plans for the small angle pitch and yaw gimbal system on the Apollo Telescope Mount (ATM) include the use of flexible pivots in place of the usually employed bearings. Because the development problems associated with a flexible pivot system operating at zero g are unknown, it was decided to test ball bearings as a backup to the flex pivot system. Outgassing ground rules for the ATM prevent the use of fluid lubricants for bearing systems. Tests made under our direction at Midwest Research Institute (ref. 1) indicate that dry film lubricants provide adequate ball bearing lubrication at low speeds and high loads. However, dry film lubricated ball bearings have not been tested previously at small angle oscillating conditions. The specific ATM oscillating gimbal system tests reported here are considered to be conceptual because bearing design and loads for gimbal bearings for the pitch and yaw system have not been defined. A size 203 bearing loaded to 120,000 psi Hertzian contact stress was chosen for the present tests. Based on this work, additional oscillating bearing tests are now being made at The Boeing Company under contract NAS8-21121 using specially designed dry lubrication systems operating through a wide range of bearing stresses.

TEST APPARATUS

The test apparatus used in this program is shown in Figures 1 and 2. This device consists of a frame on which are mounted a pair of spring loaded test bearings. The test bearings are driven by a cam through a guide bearing. The shaft between the guide bearing and the test bearings has a reduced section with a strain gage for measuring torque. The end of the bearing shaft is provided with a magnet which forms a magnetic couple with a matching magnet on the gear motor shaft outside the vacuum system. The gear motor drives the input shaft at 40 rpm which in turn oscillates the test bearing shaft through a ±5° oscillation for each rotation of the input shaft. The spring between the test bearings is designed to produce a 30 pound thrust load on the test bearings. The test apparatus is enclosed by a vacuum chamber pumped by a 75 liter/sec. ion pump which will maintain the chamber pressure at 10^{-8} torr during testing. Torque is measured by a strain gage on a torsion rod and the readings are recorded on a Sanborn Recorder.

TEST PROCEDURE

These tests are considered preliminary because no engineering data is yet available on possible bearing sizes or loads that would be required for the ATM pitch and yaw gimbal system. Size 203 bearings and a 30 pound thrust load were selected for these tests to indicate the effect of dry lubricants on oscillating bearings.

The bearing lubrication systems listed in Table I were selected for testing. A Teflon-glass ball retainer material (Duroid) (Ref. 2) was used to protect the balls from damage.

TABLE I

LUBRICATION SYSTEMS

TEST NUMBER	RETAINER MATERIAL	RACE LUBRICANT
1	Duroid	None
2	Duroid	MLF-5 MoS_2, Graphite, Gold, Sodium Silicate
3	Duroid	MLF-9 MoS_2, Graphite, Bismuth, Aluminum Phosphate
4	Duroid	MLR-2 MoS_2, Antimony Trioxide, Polyimide
5	Duroid	Vitrolube MoS_2, Glass Frit

Four test stations were utilized to test three bearing pairs for each lubrication system. Each test was operated for 100 hours or until failure of the bearings was indicated by high torque. All bearings were "run-in" for one hour at 30 rpm under full load before test. At the conclusion of this test series, the most successful lubrication system was rotated slowly through 90° to determine the friction variations which occurred during rotation due to selected spots of wear on the race lubricants.

TEST RESULTS

The first test series was made with Duroid, reinforced Teflon, retainers with no additional lubrication applied to the races. After 8 1/2 hours operation the bearing torque became extremely high and the test was shut down for inspection. Figures 3A and 3B show the heavy galling which had occurred on both the balls and races. Two additional repetitions of this test produced substantially identical results.

In the second series of tests the dry film lubricant, MLF-5 was applied to both the inner and outer bearing race. Film thickness was approximately 0.0005 inch. All three of these tests were operated the full 100 hours with a low constant torque. Inspection of the disassembled bearings at the conclusion of these tests showed no breakthrough of lubricating film and no damage to the balls. These results are shown in Figures 4A and 4B.

The third test series was made with a film thickness of 0.0005 inch of MLF-9 bonded to both races. All three tests were operated the full 100 hours of operation, but showed a slowly increasing torque during the runs. Inspection of the disassembled bearings showed no breakdown of race lubricant, but revealed that the balls were apparently damaged by the aluminum phosphate binder in the lubricant. These balls and races are shown in Figures 5A and 5B.

The fourth series of tests was made using a polyimide bonded lubricant, MLR-2, applied to the races as a film approximately 0.0003 inch thick. These tests also showed a constantly rising torque during the 100 hour period; however, as shown in Figures 6A and 6B there appeared to be no damage to either the balls or races.

The final tests were made using races coated with approximately 0.0003 inch of Vitrolube, a glass bonded MoS_2 lubricant. This lubricant is fused at over 900°F and it appears that the lubricated races were slightly warped by the high temperature. Torque readings during these tests were fairly constant with no apparent damage to the bearing as shown in Figures 7A and 7B.

Because torque values are a function of the particular bearing tested, an effective coefficient of friction was determined for all tests by determining the force in pounds at the centerline of the ball ring and dividing this force by the thrust load. Comparative friction coefficients are shown in Figure 8. Each plot on this figure represents an average of three tests.

To determine the variation in friction coefficient through a measurable distance, the test apparatus was modified to drive the loaded bearings very slowly around a 90° arc. The first test on the modified test device was made with an oil lubricated set of ball bearings loaded to 30 pounds thrust load. This test was made to determine the sensitivity of the torque measuring instrumentation and to provide a comparison with the dry film lubricated bearings. Measurements were then made on a pair of MLF-5 lubricated bearings which had been operated for 100 hours in a previous test. Results of these tests are shown in Figure 9. The following items were noted from these tests:

(a) Repetition of the bearing rotation produced identical torque patterns. This indicates that the variations in torque are caused by slight variations in film thickness and by depressions or pockets formed by the oscillating balls during previous testing. This latter theory is borne out by surface profile tests which showed that the areas of constant ball contact were approximately 6 microns lower than adjacent unworn areas in the lubricant film on the races. Pre-treatment of the lubricant film by wire brushing or polishing would probably reduce this friction variation.

(b) The low points of the friction measurements on the modified test apparatus match the friction measured during the oscillating tests of the MLF-5. This fact indicates that the higher friction coefficients are measured in the unworn or uncompacted areas of the race lubricant.

CONCLUSIONS

1. Lubrication systems are available which will provide wear lives of at least 240,000 small oscillations of approximately ±5° in ball bearings loaded to 120,000 psi Hertzian contact stress in vacuum.

2. The effective coefficient of friction of these dry lubricated ball bearings ranges from approximately 0.0055 to 0.012.

3. The maximum variation in friction coefficient around a 90° arc should not exceed .0055. This variation in friction coefficient could be reduced by pretreatment or additional run-in of the bearing lubricant before subjecting it to the oscillating condition.

4. The best lubrication system found for these test conditions includes a Teflon reinforced (Duroid) sacrificial retainer and MLF-5 coated bearing races.

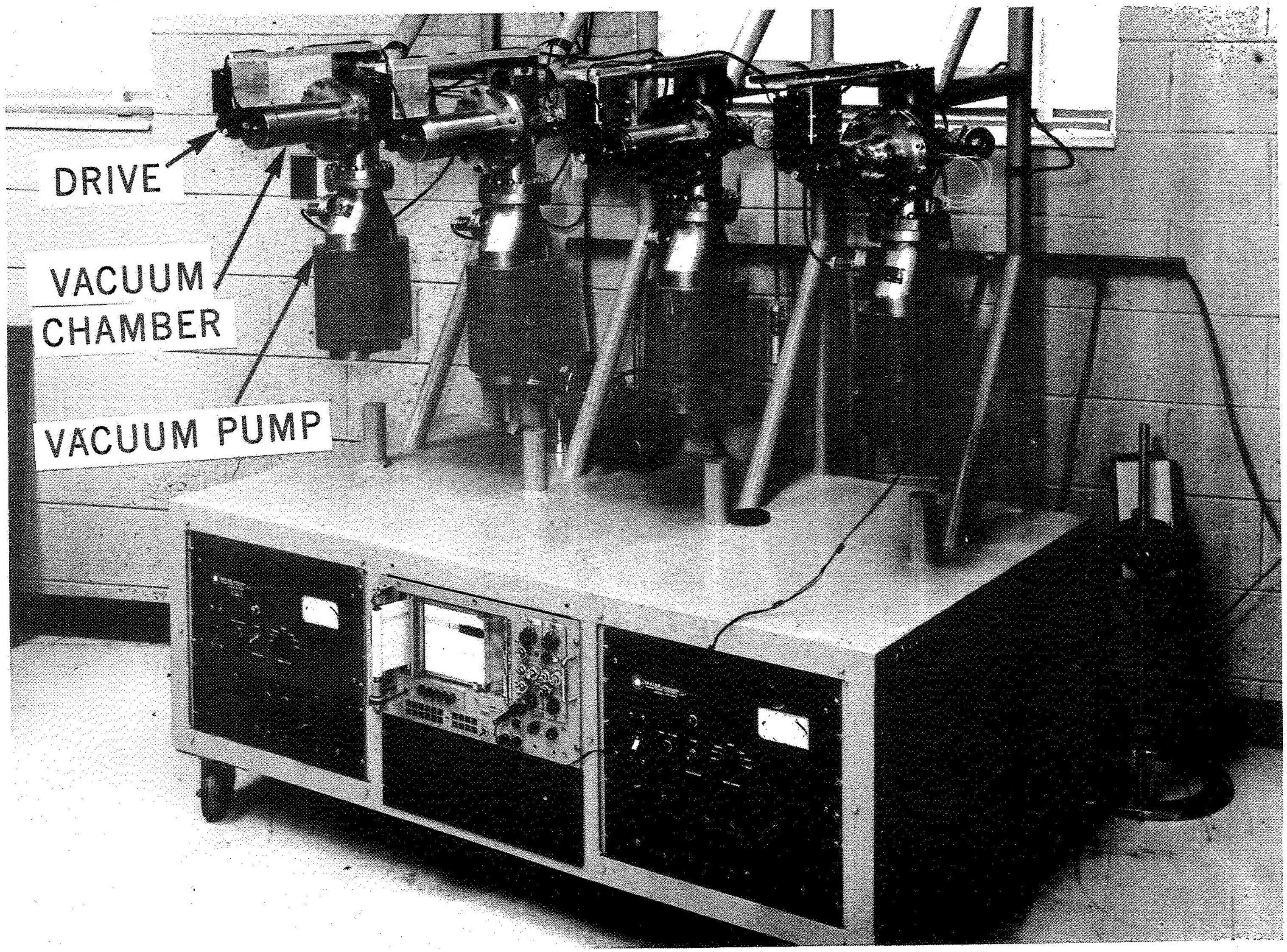


FIGURE 1. - TEST RACK ASSEMBLY

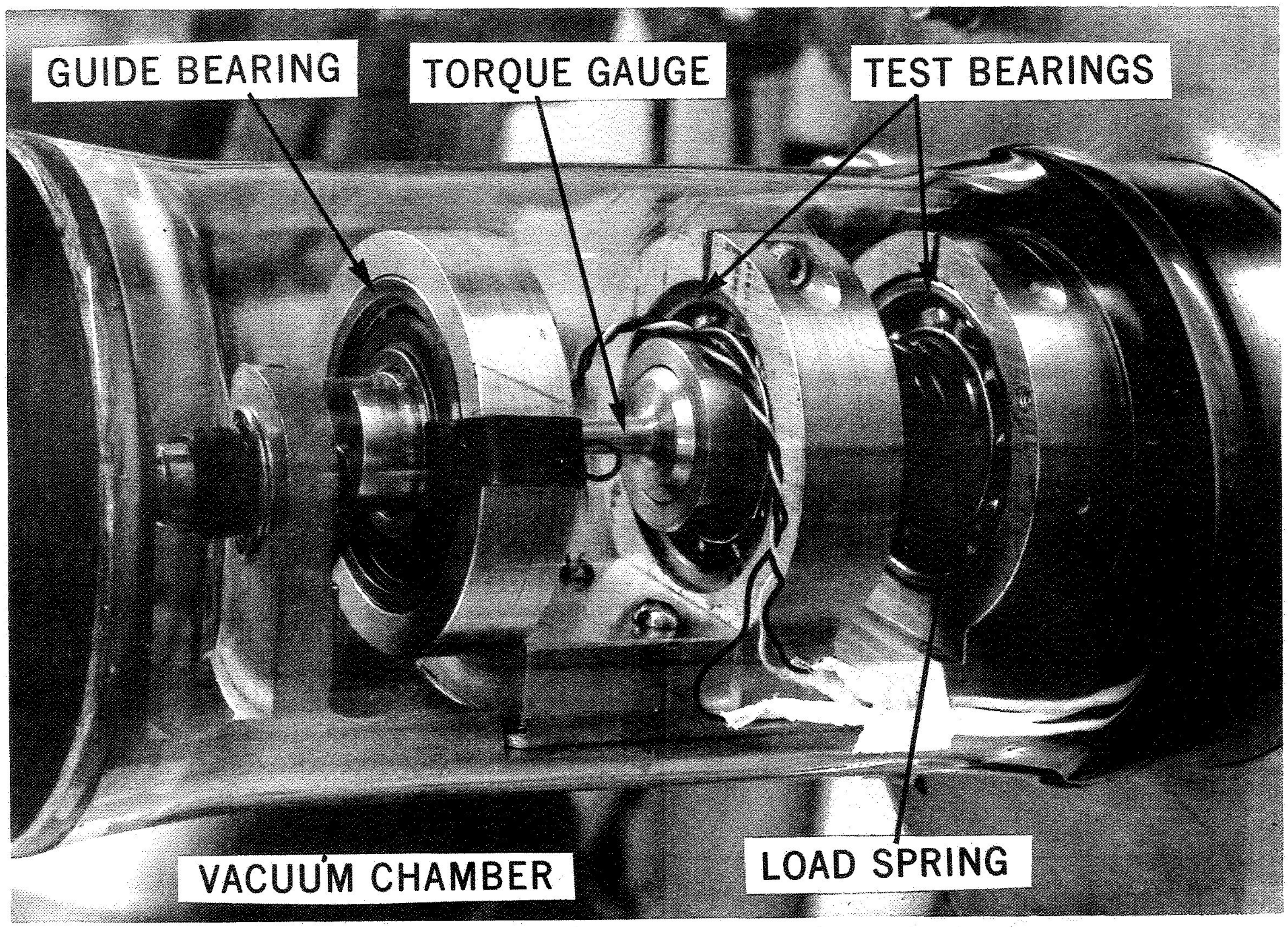


FIGURE 2. - TEST CHAMBER AND BEARING ASSEMBLY

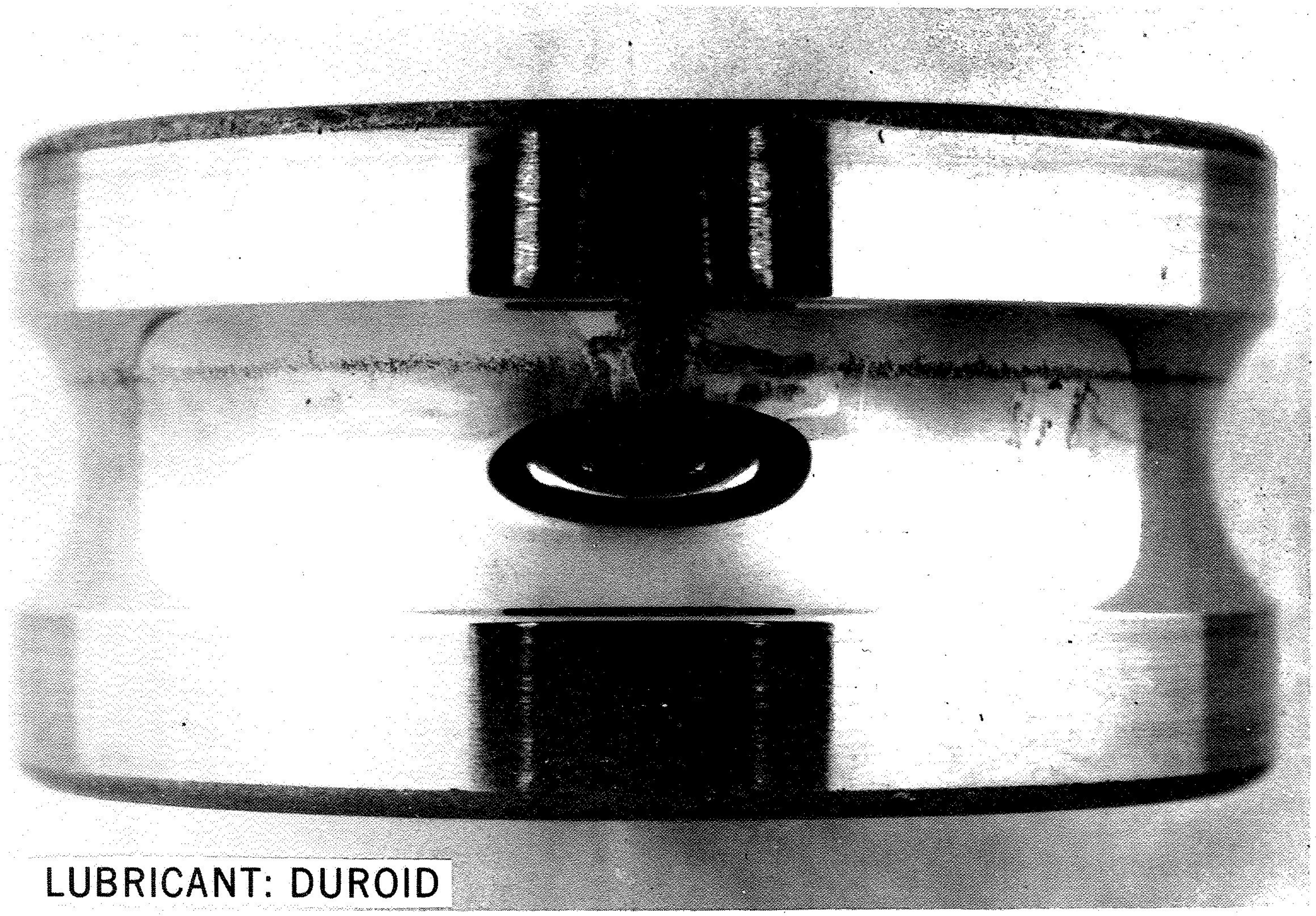


FIGURE 3A. - HEAVY GALLING ON RACE AFTER FIRST TEST SERIES

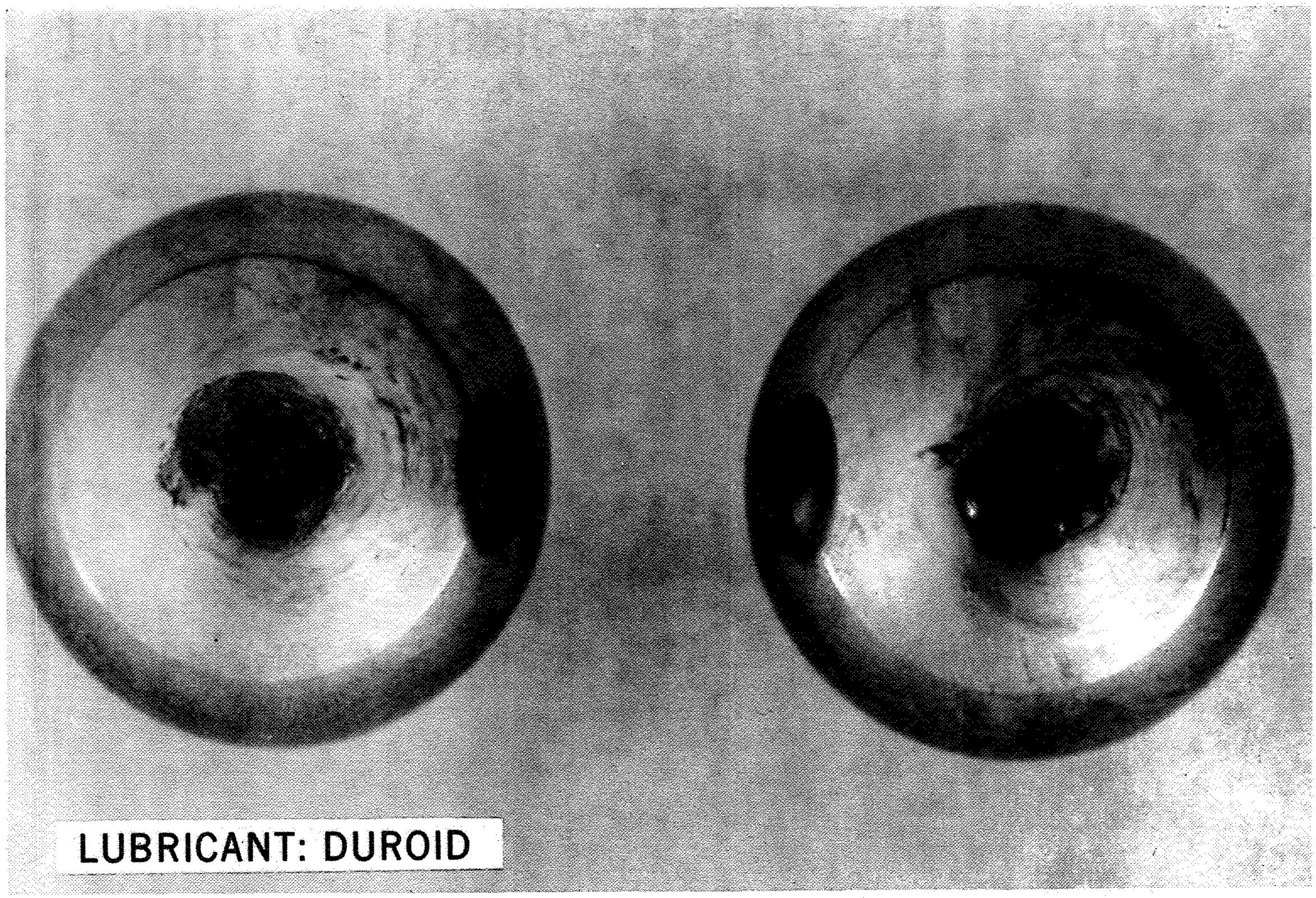


FIGURE 3B. - HEAVY GALLING ON BALLS AFTER FIRST TEST SERIES

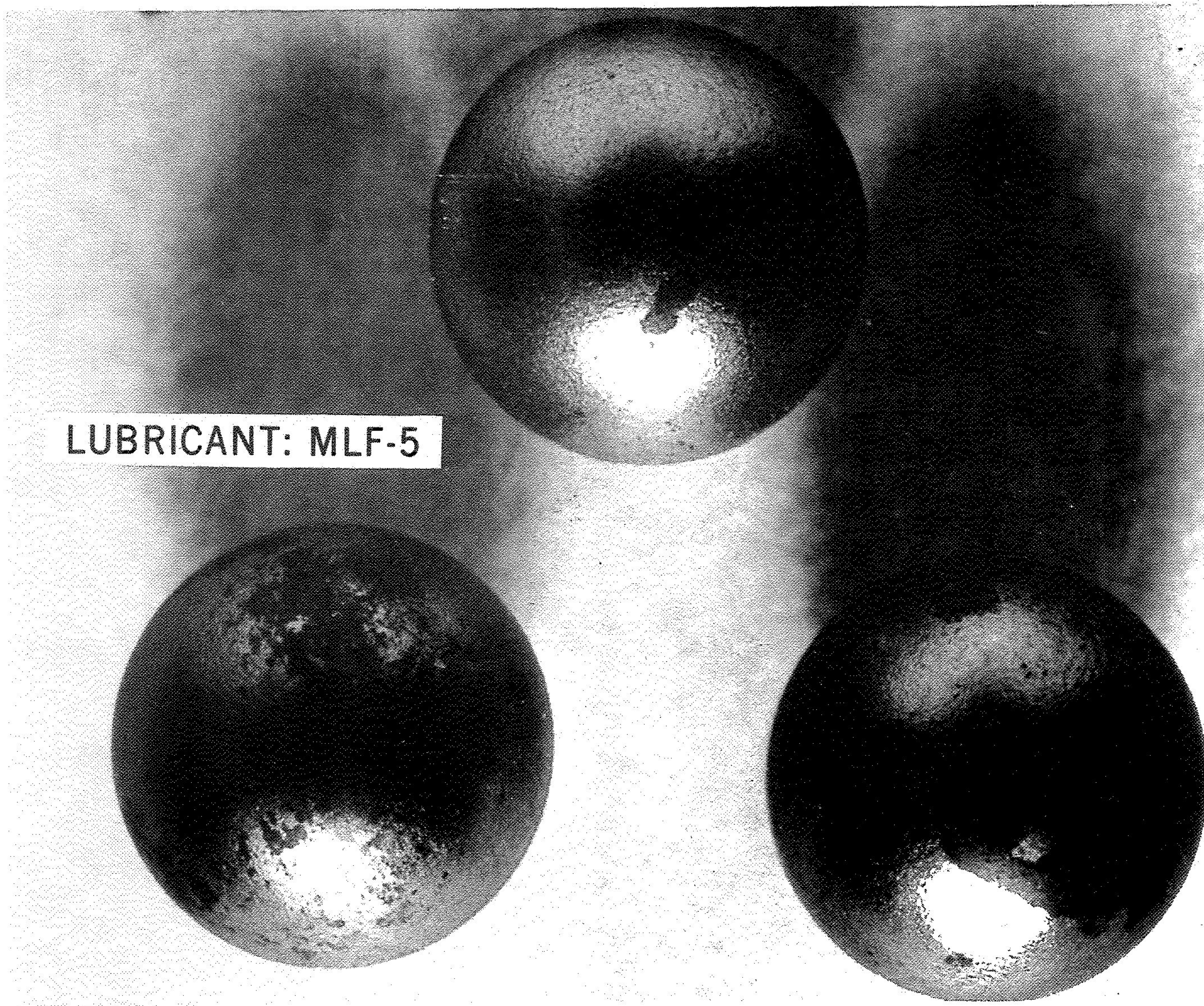
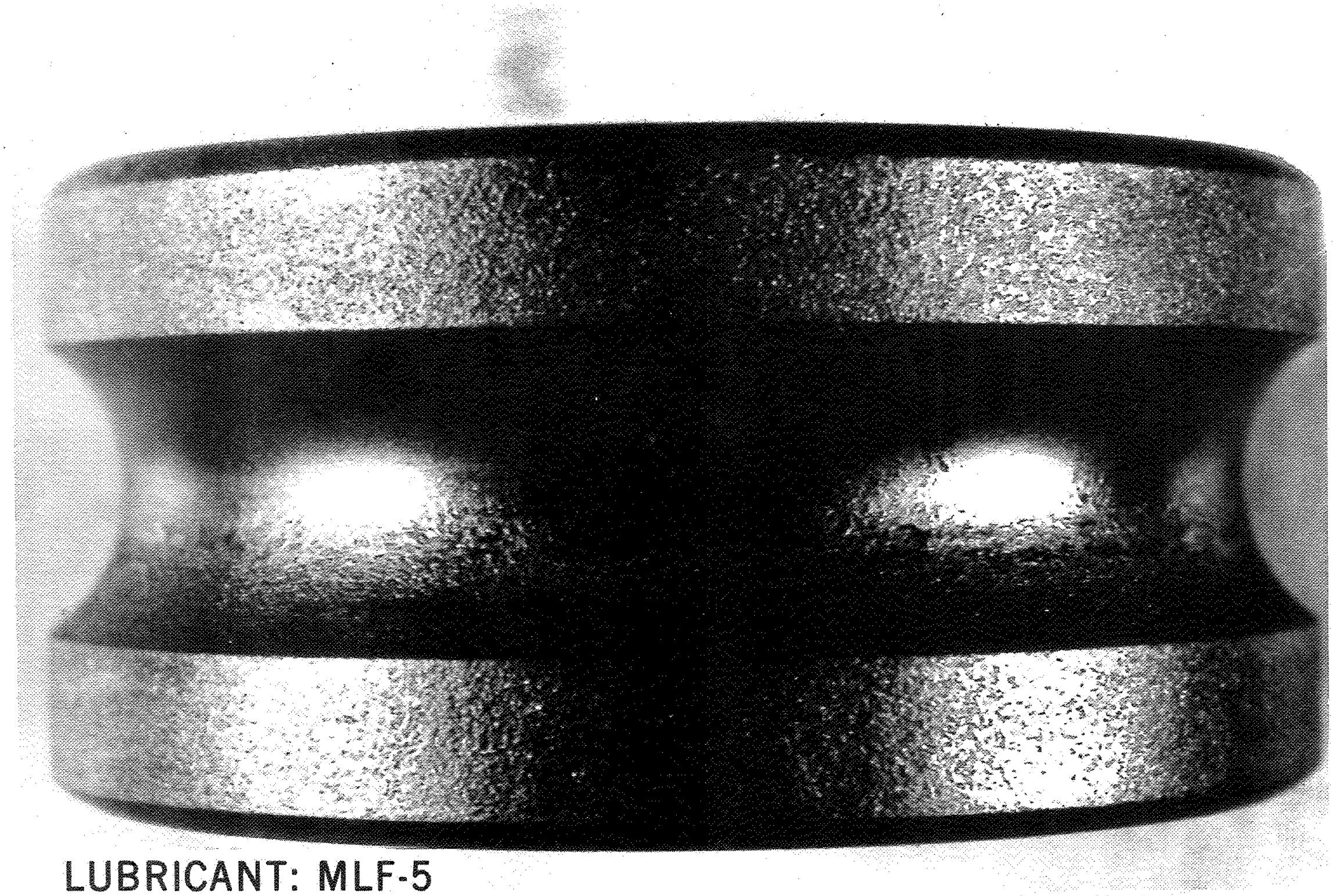
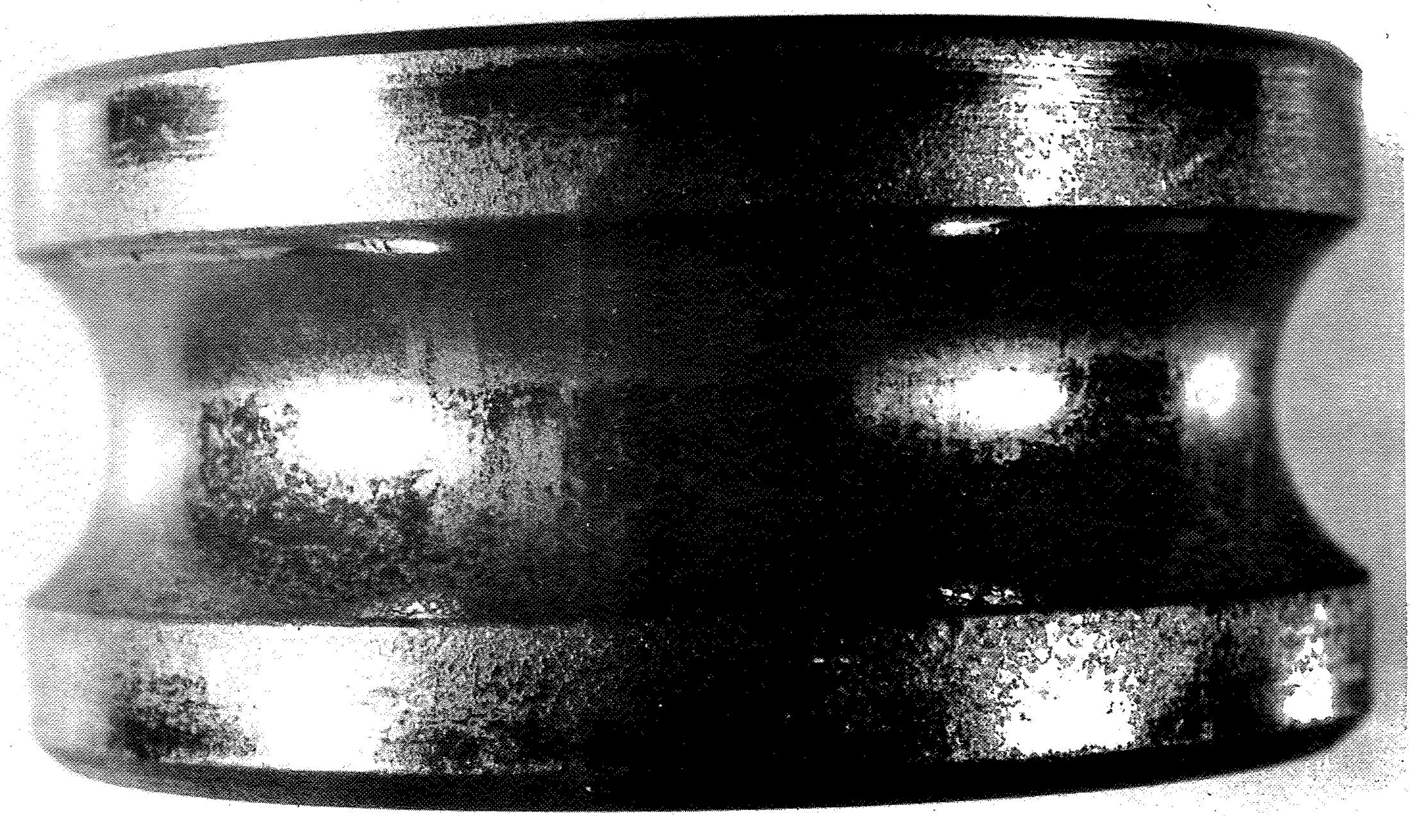


FIGURE 4A. - LUBRICATED BALLS AFTER SECOND TEST SERIES



LUBRICANT: MLF-5

FIGURE 4B. - LUBRICATED RACE AFTER SECOND TEST SERIES



LUBRICANT: MLF-9

FIGURE 5A. - RACE AT THE COMPLETION OF THIRD TEST SERIES

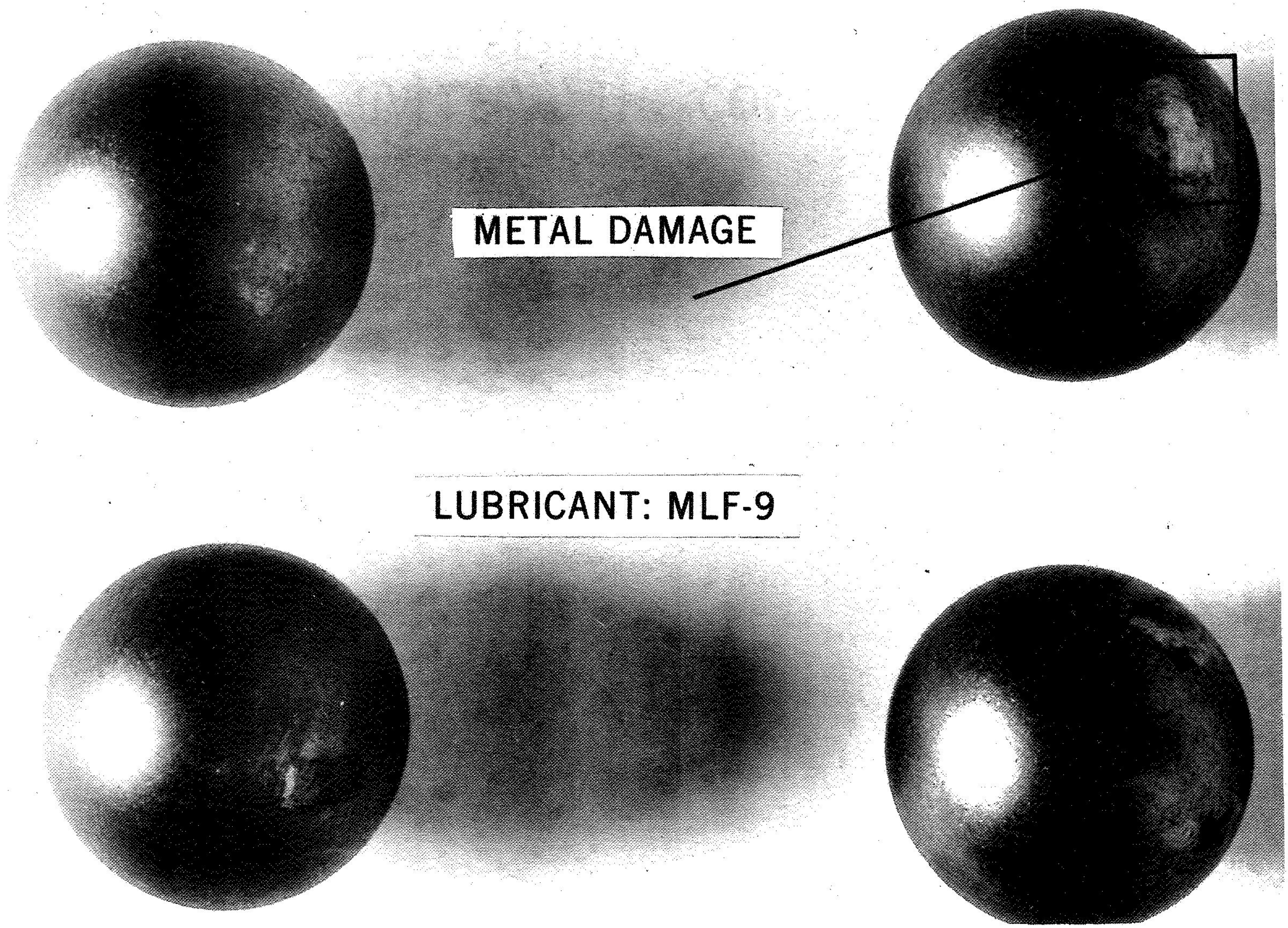


FIGURE 5B. - BALLS AT THE COMPLETION OF THIRD TEST SERIES

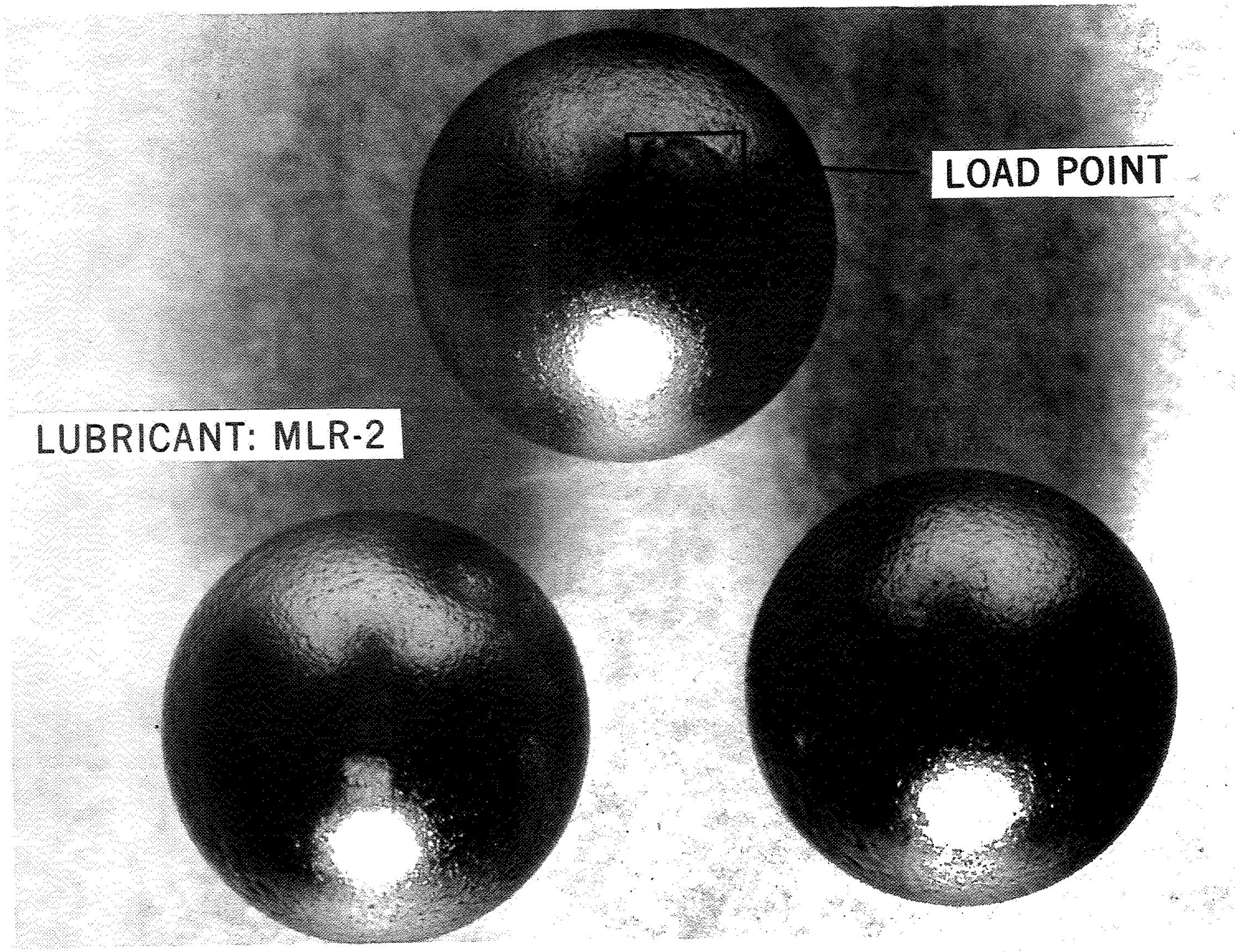


FIGURE 6A. - BALLS AT THE COMPLETION OF FOURTH TEST SERIES

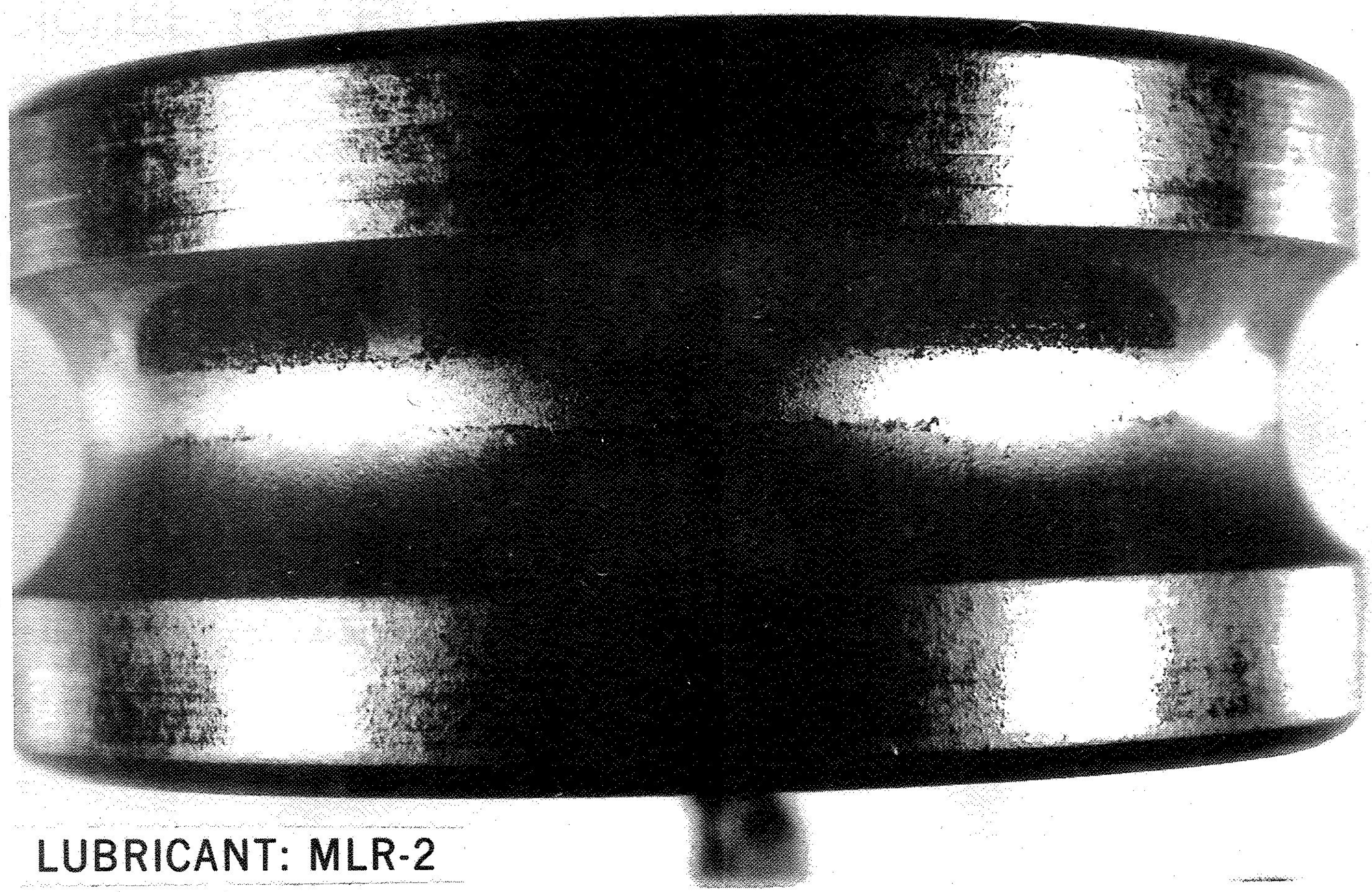


FIGURE 6B. - RACE AT THE COMPLETION OF FOURTH TEST SERIES

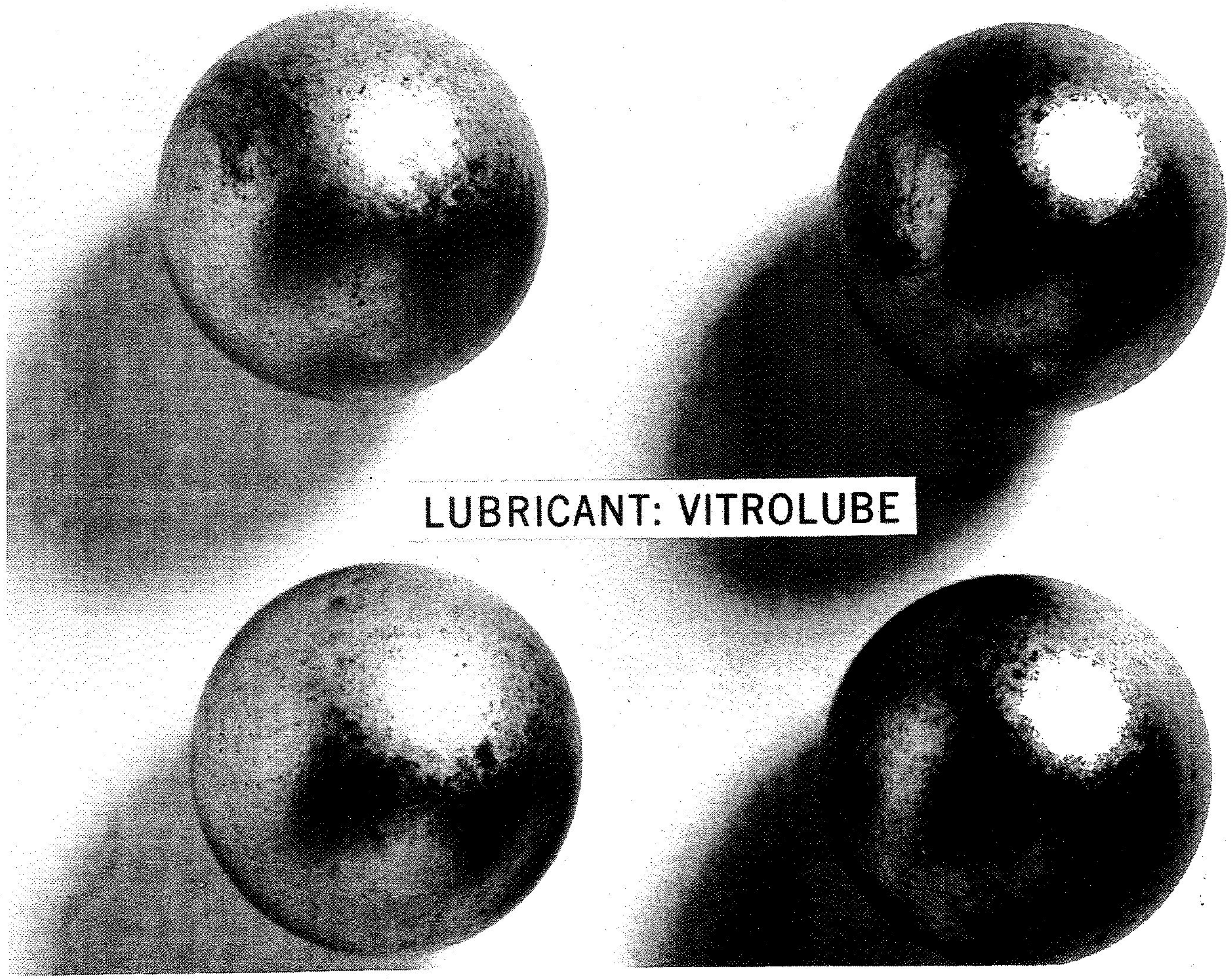
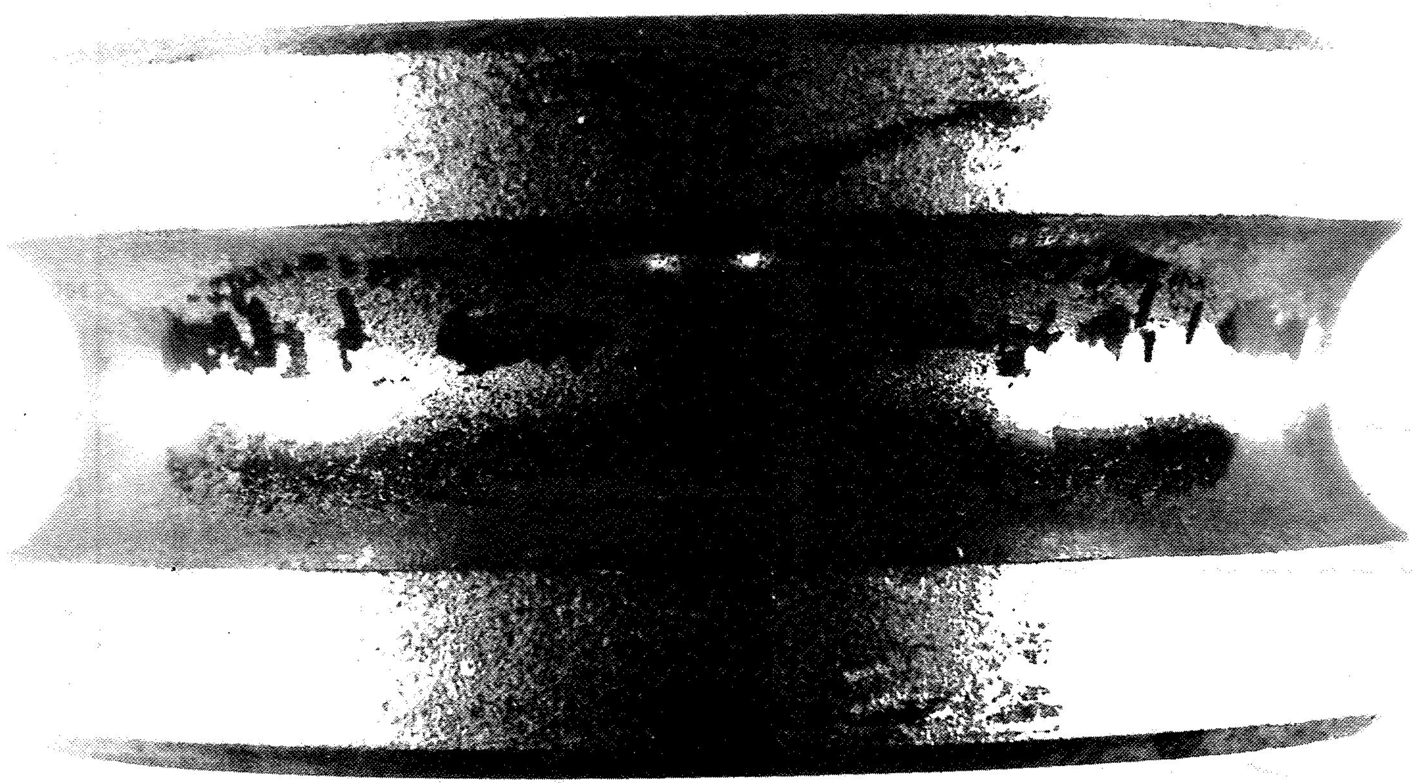


FIGURE 7A. - BALLS AT COMPLETION OF FIFTH TEST SERIES



LUBRICANT: VITROLUBE

FIGURE 7B. - RACE AT THE COMPLETION OF FIFTH TEST SERIES

MAXIMUM COEFFICIENT OF FRICTION vs TIME TYPE 203 BALL BEARINGS OSCILLIATING ± 5° AT 80 OSCILLATIONS PER MINUTE AND 30 LBS THRUST LOAD

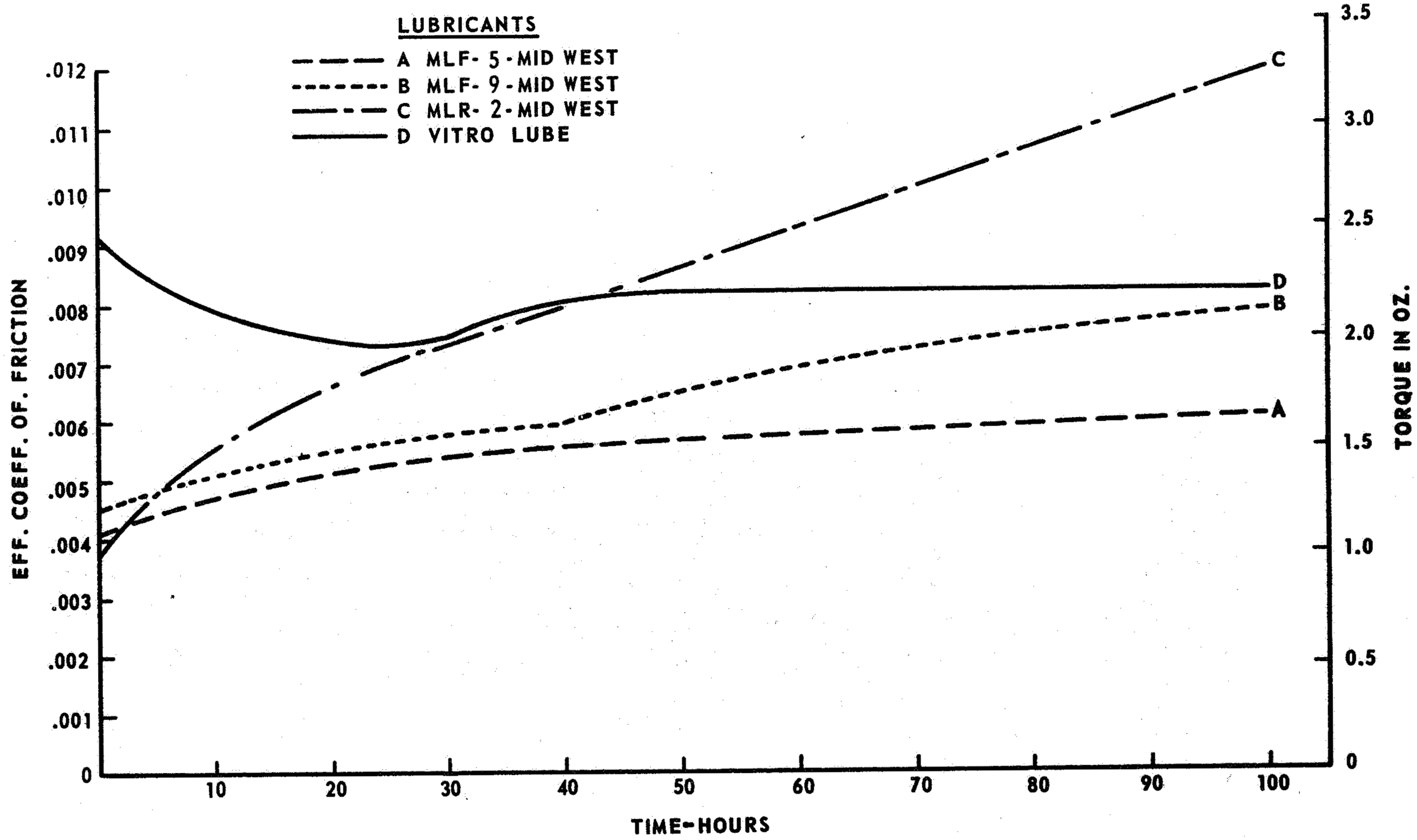


FIGURE 8. - TORQUE AND FRICTION COEFFICIENT VERSUS TIME

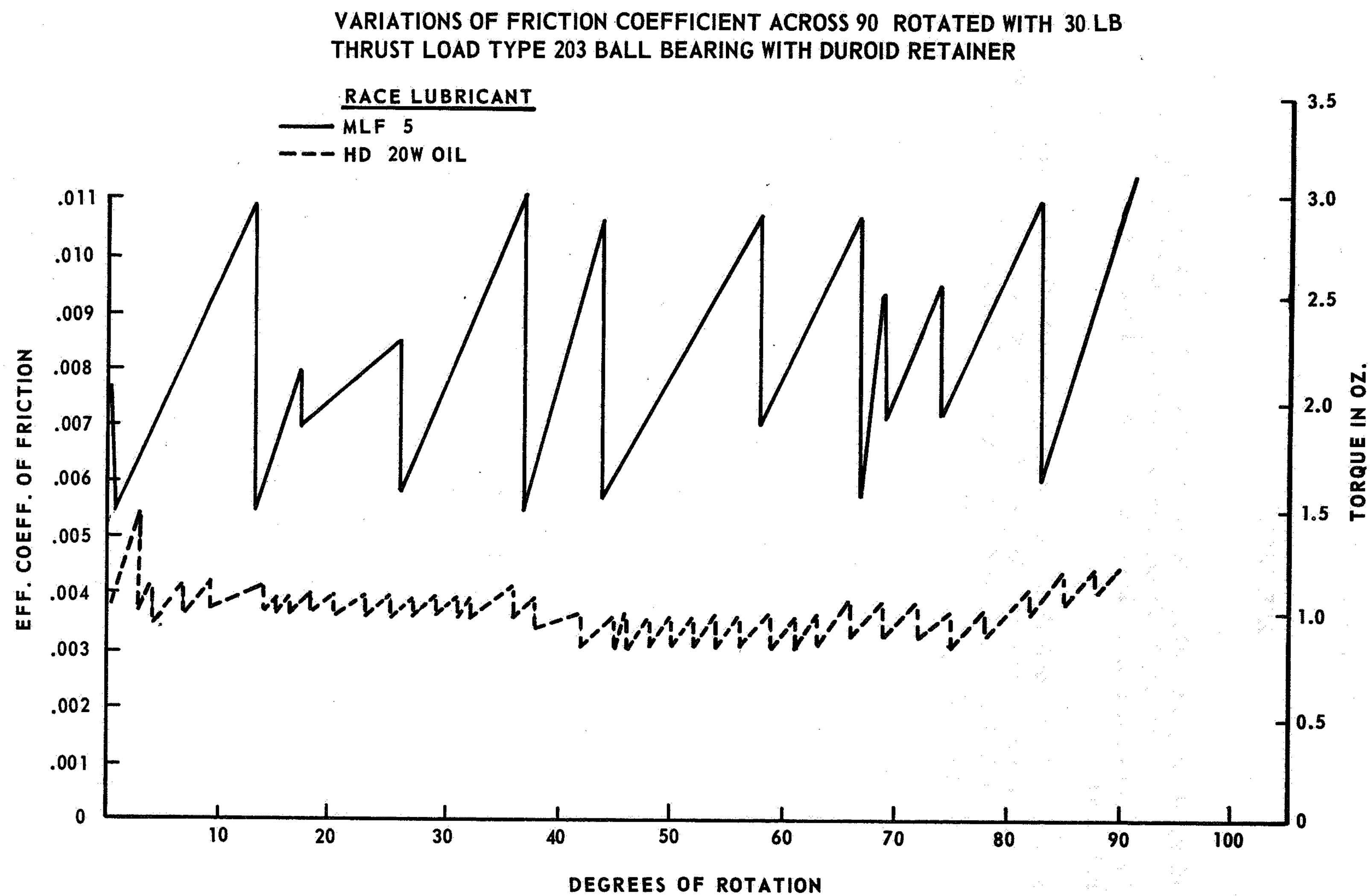


FIGURE 9. - TORQUE AND FRICTION COEFFICIENT VERSUS DEGREES OF ROTATION

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1. Gaddis, D. H., et. al., "Research on Bearing Lubricants for Use in High Vacuum," Final Summary Report of Contract NAS8-1540, Midwest Research Institute, Kansas City, Missouri, April 1967.

2. Kingsbury, J. E. and McKannan, E. C., "Materials for Space-Rating Electromechanical Components," Proceedings of the AIAA/ASME Seventh Structures and Materials Conference, Cocoa Beach, Florida, April 1966.

February 16, 1968 APPROVAL NASA TM X-53707

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The information in this report has been reviewed for security classification. Review of any information concerning Department of Defense or Atomic Energy Commission programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

This document has also been reviewed and approved for technical accuracy.

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